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CONDENSATION HEAT TRANSFER ON A HORIZONTAL CYLINDER

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~~INTRODUCTION~~ Summary

A boundary layer analysis is made for laminar film condensation on a horizontal cylinder. The formulation includes both the inertia forces and energy convection terms, which are neglected in Nusselt's simple theory. A similarity transformation, valid over most of the cylinder, is found which reduces the partial differential equations of the problem (the conservation laws) to ordinary differential equations. Numerical solutions of the resulting ordinary differential equations are available for the Prandtl number range from 0.003 to 100. Heat transfer results are presented and discussed.

NOMENCLATURE

c_p	specific heat at constant pressure
d	cylinder diameter
F	dimensionless velocity function
g	acceleration of gravity
h_{fg}	latent heat of condensation
$\bar{h}$	average heat transfer coefficient, Q/area · ΔT
k	thermal conductivity
L	height of vertical plate
Nu	overall Nusselt number; $Nu_d = \bar{h}d/k$; $Nu_L = \bar{h}L/k$
Pr	Prandtl number, $c_p\mu/k$
p	static pressure
Q	over-all heat transfer rate (per unit width)
q	local heat transfer rate per unit area
r	cylinder radius

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T	temperature; T_w, wall temperature; T_{sat}, saturation temperature of vapor; $\Delta T = T_{sat} - T_w$
u	velocity component in x direction
v	velocity component in y direction
X	dimensionless coordinate, x/r
x	coordinate measuring distance along circumference from upper stagnation point
y	coordinate measuring radial distance outward from cylinder surface
δ	thickness of condensate film
γ	function of X appearing in definition of η
η	similarity variable, $(y/r)[gr^3/\nu^2]^{1/4}\,\gamma$
θ	dimensionless temperature, $(T - T_{sat})/(T_w - T_{sat})$
μ	absolute viscosity
ν	kinematic viscosity
ρ	density
τ	shear stress
φ	function of X appearing in definition of F
ψ	stream function

INTRODUCTION

A theory of laminar film condensation was first given by Nusselt in 1916 (ref. 1). Postulating a simple model of the physical phenomenon, he successively analyzed the condensation problem for a variety of geometrical configurations. Attention is directed here to the horizontal cylinder. Our aim is to provide a modern and more complete analysis of the condensation problem, bringing it within the framework of boundary layer theory.

The physical model used by Nusselt is as follows: The hydrodynamics of the problem were expressed as a simple balance between the gravity and the shear forces, inertia effects being disregarded. With regard to the heat transfer, only heat

conduction was assumed to operate, energy convection not being considered. In the boundary layer formulation, both the inertia forces and energy convection are fully accounted for.

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The starting point of our study is the boundary layer equations appropriate to the horizontal cylinder. It will be shown that these partial differential equations can be transformed to ordinary differential equations which are valid over a major portion of the surface of the cylinder. The transformation can be carried out in such a manner that the resulting ordinary differential equations coincide with those for condensation on a vertical flat plate. Utilizing numerical solutions of the transformed equations, heat transfer results are presented for the horizontal cylinder over the Prandtl number range from 0.003 to 100.

ANALYSIS

The conservation equations. - The system under consideration is shown in schematic view in figure 1. The x-coordinate measures the distance along the circumference of the cylinder, while y measures the distance normal to the surface. It will be supposed that the cylinder is situated in a large body of quiescent, pure vapor which is at its saturation temperature T_{sat}. The surface temperature of the cylinder, T_w, is uniform and $T_w < T_{sat}$. A continuous film of condensate runs downward over the cylinder.

The condensation problem is governed by the basic conservation principles: mass, momentum, and energy. The boundary layer form of these laws for constant property, nondissipative, gravity flow over a horizontal cylinder is

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$\rho\left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y}\right) = \rho g \sin \frac{x}{r} + \mu \frac{\partial^2 u}{\partial y^2} \tag{2}$$

$$\rho c_p\left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y}\right) = k \frac{\partial^2 T}{\partial y^2} \tag{3}$$

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The static pressure gradient, $\partial p/\partial x$, has been neglected in equation (2) relative to the gravity force, $\rho g \sin x/r$.

The conservation of mass equation (2) is immediately satisfied by defining the stream function ψ in the usual way, i.e.,

$$u = \partial\psi/\partial y, \qquad v = -\partial\psi/\partial x \tag{4}$$

Then, the velocity components u and v appearing in equations (2) and (3) are replaced in favor of ψ, giving

$$\frac{\partial\psi}{\partial y}\frac{\partial^2\psi}{\partial x\partial y} - \frac{\partial\psi}{\partial x}\frac{\partial^2\psi}{\partial y^2} = g\sin(x/r) + \nu\frac{\partial^3\psi}{\partial y^3} \tag{2a}$$

$$\frac{\partial\psi}{\partial y}\frac{\partial T}{\partial x} - \frac{\partial\psi}{\partial x}\frac{\partial T}{\partial y} = \frac{k}{\rho c_p}\frac{\partial^2 T}{\partial y^2} \tag{3a}$$

Faced with this formidable pair of partial differential equations, we are motivated to seek some means for reducing them to ordinary differential equations, which are easier to solve.

Reduction to ordinary differential equations. - Guided by previous experience with boundary layer theory, we propose a new independent variable η as follows

$$\eta = \frac{y}{r}\left[\frac{gr^3}{\nu^2}\right]^{1/4} \gamma(X) \tag{5a}$$

where γ is a function of X which is not yet determined. In addition, new dependent variables are constructed in the following way

$$F(\eta) = \psi\Big/\nu\left[\frac{gr^3}{\nu^2}\right]^{1/4} \varphi(X), \qquad \theta(\eta) = \frac{T - T_{sat}}{T_w - T_{sat}} \tag{5b}$$

where $\varphi(X)$ is also undetermined. The variable θ is seen to be a dimensionless temperature, while F is related to the velocities of the problem by the equations

$$u = \frac{\nu}{r}\left[\frac{gr^3}{\nu^2}\right]^{1/2} \gamma\varphi F' \tag{5c}$$

$$v = -\frac{\nu}{r}\left[\frac{gr^3}{\nu^2}\right]^{1/4}\left[\frac{d\varphi}{dX}F + \frac{\varphi(d\gamma/dX)}{\gamma}\eta F'\right] \tag{5d}$$

The primes denote differentiation with respect to η. Inspection of equations (5a) through (5d) reveals that the structures of the variables η, F, and θ are those normally found in similar-type boundary layer formulations.

Using these new variables, a formal transformation of equations (2a) and (3a) may be carried out. The end result of this operation is

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$$F'''(\varphi\gamma^3) + FF''\left(\varphi \frac{d\varphi}{dX} \gamma^2\right) - (F')^2\left[\varphi \frac{d\varphi}{dX} \gamma^2 + \varphi^2\gamma \frac{d\gamma}{dX}\right] = - \sin X \tag{6}$$

$$\theta'' + Pr\ \theta'\left(\frac{1}{\gamma} \frac{d\varphi}{dX}\right) = 0 \tag{7}$$

In order that equations (6) and (7) be ordinary differential equations, it is required the factors which depend on X be eliminated. Specifically, it is necessary that

$$\varphi\gamma^3 = a \sin X \tag{8a}$$

$$\varphi \frac{d\varphi}{dX} \gamma^2 = b \sin X \tag{8b}$$

$$\varphi^2\gamma \frac{d\gamma}{dX} = c \sin X \tag{8c}$$

$$\frac{d\varphi}{dX} = e\gamma \tag{8d}$$

where a, b, c, and e are constants. Thus, there is seen to be a set of <u>four</u> equations for determining the <u>two</u> functions γ and φ. In order that two functions be determined by four equations, it is required that the equations are not independent of each other; but rather, that two of the equations follow from the other two. Provided that a unique solution can be found, then equations (6) and (7) will truly be ordinary differential equations and the transformation defined by equations (5a) and (5b) will have achieved its end.

The problem of finding φ and γ from equations (8a) through (8d) is precisely the same one faced by Hermann in analyzing free convection about a horizontal cylinder (ref. 2). He found that a unique solution could be achieved except for small values of X. However, the boundary layer assumptions are already invalid for small X for other reasons. In view of this, it has been generally accepted that the failure

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to find a unique solution for φ and γ in this region will not introduce errors in the over-all heat transfer significantly different from those already present in the boundary layer analysis.

The functions φ and γ as given by Hermann are listed in Table I.*

TABLE I. - φ AND γ FUNCTIONS

X	0°	30°	60°	90°	120°	150°	165°	180°
φ	0	1.181	2.337	3.430	4.41	5.24	5.55	5.84
γ	0.760	.752	.718	.664	.581	.458	.360	0

The level of these functions has been adjusted so that the constants a, b, c, and e of equations (8a) through (8d) are simple integers which will lead to a particularly desirable form when inserted in equations (6) and (7). The γ function and the product $\gamma\varphi$, originally presented as figure 17 of reference 2, have been reproduced in figure 2.

Returning now to equations (6) and (7) and utilizing the φ and γ functions, there results

$$F''' + 3FF'' - 2(F')^2 + 1 = 0 \tag{9a}$$

$$\theta'' + 3(Pr)F\theta' = 0 \tag{9b}$$

It is of first importance to observe that these equations coincide precisely with the governing equations for film condensation on a vertical plate (ref. 3). (This correspondence was assured by adjustment of the levels of φ and γ.) The boundary conditions on F and θ, to be discussed below, are also the same as for the vertical plate problem. As a consequence, the available mathematical solutions for the F,θ of the vertical plate are immediately at our disposal as solutions for the F,θ of the horizontal cylinder. However, since different transformations were used, the physical variables (such as heat transfer, velocity, and temperature) do not follow the same laws in the two problems.

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*In reference 2, the φ function is denoted as f, while the γ function is denoted as g.

The boundary conditions. - To complete the statement of the problem, the boundary conditions must be specified. At the surface of the cylinder, both velocity components must vanish as a consequence of the no-slip condition of viscous flow and of the impermeability of the wall. Also, the condensate immediately adjacent to the wall must take on the temperature T_w. At the interface between the liquid and vapor ($y = \delta$), the condition is imposed that the vapor offers negligible shearing resistance to the condensate flow. This assumption is standard in condensation theory. In addition, it will be required that the condensate take on the temperature T_{sat} at the liquid-vapor interface. The formal statement of these conditions is

$$\left.\begin{aligned} u &= 0 \\ v &= 0 \\ T &= T_w \end{aligned}\right\} y=0 \qquad \left.\begin{aligned} \tau &= \mu\, \partial u/\partial y = 0 \\ T &= T_{sat} \end{aligned}\right\} y = \delta \tag{10}$$

The thickness of the condensate layer δ, which will vary with position around the cylinder, must be determined from the analysis. In terms of the new variables of equations (5a) through (5d), these boundary conditions become

$$\left.\begin{aligned} F &= 0 \\ F' &= 0 \\ \theta &= 1 \end{aligned}\right\} \eta = 0 \qquad \left.\begin{aligned} F'' &= 0 \\ \theta &= 0 \end{aligned}\right\} \eta = \eta_\delta \tag{10a}$$

where η_δ corresponds to the value of η at $y = \delta$.

As a consequence of the transformation (5), our problem has been reduced to the task of solving equations (9) subject to the boundary conditions (10a). Evidently, these solutions will depend on the choice of two parameters: Pr and η_δ. From the mathematical standpoint, this is a completely satisfactory situation. But, practically speaking, the problem is incomplete, since the dimensionless condensate layer thickness η_δ would not be know a priori. So, there remains the need to relate η_δ to another parameter which contains physical quantities which are all known.

To relate η_δ (and hence δ) to known physical quantities, we invoke an over-all energy balance as follows

$$h_{fg}\int_0^\delta \rho u \, dy + \int_0^\delta \rho u c_p(T_{sat} - T)dy = \int_0^x k\left(\frac{\partial T}{\partial y}\right)_{y=0} dx \tag{11}$$

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The first term on the left is the energy released as latent heat, while the second term is the energy liberated by subcooling of the condensate. The right hand side represents the heat transferred from the condensate to the cylinder over a span of circumference from $x = 0$ to $x = x$. In writing equation (11), the standard assumption of negligible heat conduction across the liquid-vapor interface has been invoked. In terms of the transformed variables as given by equation (5), the over-all energy balance becomes

$$\frac{c_p\Delta T}{h_{fg}} = -3\mathrm{Pr}\ \frac{F(\eta_\delta)}{\theta'(\eta_\delta)} \tag{11a}$$

where $F(\eta_\delta)$ and $\theta'(\eta_\delta)$ respectively denote the values of F and $d\theta/d\eta$ at $\eta = \eta_\delta$.

From a solution of equations (9) corresponding to specified values of the parameters Pr and η_δ, the values of $F(\eta_\delta)$ and $\theta'(\eta_\delta)$ are available, and the right side of equation (11a) may be evaluated. So, $c_p\Delta T/h_{fg}$ is determined. For a fixed Prandtl number, equations (9) may be solved for a sequence of values of η_δ; and a corresponding set of values of $c_p\Delta T/h_{fg}$ may then be computed from equation (11a). In other words, for a fixed Prandtl number, there is a unique relation between η_δ and $c_p\Delta T/h_{fg}$. So, we may regard the solutions as depending on the parameters Pr and $c_p\Delta T/h_{fg}$, rather than on Pr and η_δ.

Solutions. - It has already been remarked that equations (9) and the boundary conditions (10a) coincide with those for condensation on a vertical flat plate. Further, the relationship between $c_p\Delta T/h_{fg}$ and η_δ for a fixed Prandtl number, as given by equation (11a), is the same as that for the plate. So, immediate application of the mathematical solutions for the plate can be made to the horizontal cylinder.

For the vertical plate, solutions of equations (9) subject to the boundary conditions (10a) have been carried out numerically in reference 3 for Prandtl numbers of 0.003, 0.008, 0.03, 1.0, 10, and 100. The first three of these Prandtl numbers correspond to liquid metals, while the last three correspond to ordinary liquids. Utilizing these numerical solutions, heat transfer results for the horizontal cylinder will be reported below for values of the parameter $c_p\Delta T/h_{fg}$ ranging from essentially 0 to 1.

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RESULTS

Heat transfer. - The quantity of greatest practical interest is the heat transferred from the condensate to the cylinder. The local heat flux is computed from Fourier's Law, i.e.,

$$q = k\left(\frac{\partial T}{\partial y}\right)_{y=0} \tag{12}$$

In terms of the transformed variables of equation (5), the expression for q becomes

$$q = \frac{k}{r}\,(T_{sat} - T_w)\left[-\theta'(0)\right]\left(\frac{gr^3}{\nu^2}\right)^{1/4} \gamma(X) \tag{12a}$$

where $\theta'(0)$ is an abbreviation for $(d\theta/d\eta)_{\eta=0}$. The variation of q around the circumference is seen to be proportional to the variation of the γ function. Using figure 2, it may be observed that q has its maximum value at $X = 0$ (upper stagnation point) and decreases monotonically with increasing X.

The over-all heat transfer to the entire cylinder may be found by integrating the local values according to the relation

$$Q = 2\int_0^{\pi r} q\,dx = 2r\int_0^{\pi} q\,dX \tag{13}$$

Introducing q from equation (12a) and integrating, there is obtained

$$Q = 2k(T_{sat} - T_w)\left[-\theta'(0)\right]\left(\frac{gr^3}{\nu^2}\right)^{1/4} \int_0^{\pi} \gamma\,dx \tag{13a}$$

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The value of the integral is given by Hermann as

$$\int_0^{\pi} \gamma \, dX = 0.616 \, \pi$$

It is customary to phrase the heat transfer results in terms of an over-all heat transfer coefficient and an over-all Nusselt number defined as follows

$$\bar{h} = \frac{Q}{2\pi r(T_{sat} - T_w)}, \qquad Nu_d = \frac{\bar{h}d}{k} \tag{14}$$

With this, the over-all heat transfer result of equation (13a) becomes

$$Nu_d \bigg/ \left(\frac{gd^3}{\nu^2}\right)^{1/4} = 0.733 \left[-\theta'(0)\right] \tag{15}$$

Since θ (0) depends upon both Pr and $c_p\Delta T/h_{fg}$, so then does the dimensionless heat transfer representation of equation (15). Previous experience with condensation heat transfer theory suggests that the dependence on Pr and $c_p\Delta T/h_{fg}$ can be substantially reduced by rephrasing the dimensionless heat transfer in the form

$$Nu_d \bigg/ \left[\frac{g\rho h_{fg} d^3}{k\nu(T_{sat} - T_w)}\right]^{1/4} = 0.733 \left[-\theta'(0)\right] \left[\frac{Pr}{c_p\Delta T/h_{fg}}\right]^{1/4} \tag{16}$$

Utilizing the numerical solutions of equations (9), the right hand sides of equations (16) and (11a) can be evaluated. The results thus obtained have been plotted on figure 3.* This graph contains the complete theoretical prediction of the heat transfer as given by boundary layer theory.

By inspection of the figure, it is immediately seen that for small values of $c_p\Delta T/h_{fg}$, the heat transfer results for fluids of any Prandtl number are given by

$$Nu_d = 0.733 \left[\frac{g\rho h_{fg} d^3}{k\nu(T_{sat} - T_w)}\right]^{1/4} \tag{17}$$

Now, it has been shown in reference 3 that the thickness of the condensate film varies monotonically with $c_p\Delta T/h_{fg}$. So, small values of $c_p\Delta T/h_{fg}$ correspond to thin

*The curves for liquid metals have been extended only to $c_p\Delta T/h_{fg} = 0.1$ in consideration of the relatively small magnitudes of c_p/h_{fg} possessed by these fluids.

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films. In very thin condensate films, the effects of both energy convection and inertia forces are very small. So, equation (17) describes the heat transfer process in the limiting situation where the energy convection and inertia play a negligible role. It is interesting to observe that Nusselt's analysis, which deals only with this limiting situation, leads to a result identical to equation (12), except that his constant is 1 percent lower.

Relative to the predictions of equation (17), the role of energy convection would be to increase the heat transfer, while the role of the inertia forces would be to decrease the heat transfer. So, these two effects are in conflict. With this, we can provide perspective for the trends displayed on figure 3. For the high Prandtl number fluids, where the inertia forces are completely overriden by the viscous forces, the effects of energy convection win out and the curves rise with increasing film thickness (increasing $c_p\Delta T/h_{fg}$). For the low Prandtl number fluids, where the heat conductivity overrides the energy convection, the effects of the inertia forces win out and the curves fall with increasing film thickness.

Heat transfer results for small cylinders. - For a given temperature condition and a given fluid (i.e., given $c_p\Delta T/h_{fg}$ and Pr), our theory predicts that

$$\eta_\delta = \frac{\delta}{r}\left[\frac{gr^3}{\nu^2}\right]^{1/4} \gamma(X) = \text{constant} \tag{18}$$

or

$$\frac{\delta}{r} \sim \frac{1}{r^{3/4}\gamma(X)} \tag{18a}$$

At any circumferential position X, it is seen that the ratio δ/r will increase as the cylinder radius decreases. Since the boundary layer equations for the horizontal cylinder are based on the assumption that δ/r is small, it follows that boundary layer theory fails for cylinders of very small radius.

The boundary layer heat transfer results may be corrected for the effects of small radius in a manner identical to that used by Eckert (ref. 4, p. 165) for the

free convection problem. Translating his findings into the context of the condensation problem, there is obtained

$$Nu_d' = \frac{2}{\ln\left(1 + \frac{2}{Nu_d}\right)} \tag{19}$$

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where Nu_d' is the corrected Nusselt number and Nu_d is the prediction for full-sized cylinders (e.g., fig. 3). For normal applications, $2/Nu_d \ll 1$, so

$$\ln\left(1 + \frac{2}{Nu_d}\right) \approx \frac{2}{Nu_d}, \qquad Nu_d' \approx Nu_d$$

Comparison with vertical plate heat transfer. - For film condensation on a vertical plane of height L, the dimensionless heat transfer prediction of boundary layer theory takes the form

$$Nu_L \Bigg/ \left[\frac{g\rho h_{fg} L^3}{k\nu(T_{sat} - T_w)}\right]^{1/4} = 0.943\left[-\theta'(0)\right]\left[\frac{Pr}{c_p\Delta T/h_{fg}}\right]^{1/4} \tag{20}$$

where $\theta'(0)$ is the same function of Pr and $c_p\Delta T/h_{fg}$ as appears in equation (16) and

$$Nu_L = \frac{\bar{h}L}{k}, \qquad \bar{h} = \frac{Q}{L(T_{sat} - T_w)}$$

So, for a given Pr and $c_p\Delta T/h_{fg}$, it follows that

$$Nu_L \Bigg/ 0.943\left[\frac{g\rho h_{fg} L^3}{k\nu(T_{sat} - T_w)}\right]^{1/4} = Nu_d \Bigg/ 0.733\left[\frac{g\rho h_{fg} d^3}{k\nu(T_{sat} - T_w)}\right]^{1/4} \tag{21}$$

As a consequence of this correspondence, figure 3 can be used equally well for the flat plate as for the horizontal cylinder.

It is interesting to inquire about the relationship between plate height and cylinder diameter required to produce equal heat transfer coefficients for plate and cylinder. Equating heat transfer coefficients, temperature differences, and fluid properties in equation (21), it is found that

$$L = 2.78\ d$$

Using this equivalence, correlations of the heat transfer coefficient for cylinders may also be used for vertical plates and vice-versa.

Circumferential variation of film thickness. - For a particular physical situation (i.e., given fluid, ΔT and r), the circumferential variation of the film thickness δ is found from equation (18) to be

$$\delta \sim \frac{1}{\gamma(X)} \tag{22}$$

The function $\gamma(X)$ is graphed on figure 2, and its reciprocal provides us with the variation of δ. Utilizing figure 2, it is seen that the condensate layer thickness, starting from a finite value at $X = 0$, grows rather slowly over the upper portion of the cylinder and then increases quite rapidly over the lower part of the cylinder. At the lower stagnation point $(X = \pi)$, the theoretical prediction of an infinite film thickness can be interpreted as signifying the dropping away of the condensate from the cylinder surface.

Comparison with experimental heat transfer. - Experiments on film condensation over a horizontal cylinder have been carried out primarily for fluids with Prandtl numbers lying above unity. Data for these high Prandtl number experiments are summarized by McAdams (ref. 5, p. 340). In general, the conditions of the tests corresponded to small values of $c_p\Delta T/h_{fg}$. Since the analytical predictions of Nusselt closely correspond to those of the boundary layer formulation for low $c_p\Delta T/h_{fg}$, the comparison of McAdams (table 13-4) between experimental data and Nusselt's theory also applies here.

For the low Prandtl number range, experiments on sodium and mercury have been carried out Misra and Bonilla for values of $c_p\Delta T/h_{fg}$ up to 0.03. Their heat transfer results were only 5 to 15 percent of equation (17). While boundary layer theory predicts a lowering of the Nusselt number for these low Prandtl number fluids (see fig. 3), the predicted reduction is much smaller than that found by experiment. So, it would appear that the effect of the inertia forces is by no means sufficient to explain the experimental findings. There still remains the need for further experiments to clearly define the departures between the test conditions and the analytical model.

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REFERENCES

1. W. Nusselt, "Die Oberflächen Kondensation des Wasserdampfes," Zeitschrift des Vereines Deutscher Ingenieure, vol. 60, 1916, p. 541 and 569.

2. R. Hermann, "Heat Transfer by Free Convection from Horizontal Cylinders in Diatomic Gases," NACA TM 1366, 1954.

3. E. M. Sparrow and J. L. Gregg, "A Boundary Layer Treatment of Laminar Film Condensation," ASME paper, no. 58-SA-2. To appear in Trans. ASME.

4. E. R. G. Eckert, "Introduction to the Transfer of Heat and Mass," First edition, McGraw-Hill Book Co., New York, N. Y., 1950.

5. William H. McAdams, "Heat Transmission," Third edition, McGraw-Hill Book Co., New York, N. Y., 1954.

6. B. Misra and C. F. Bonilla, "Heat Transfer in the Condensation of Metal Vapors," Chem. Eng., Progress Symposium Series no. 18, vol. 52, pp. 7-21.

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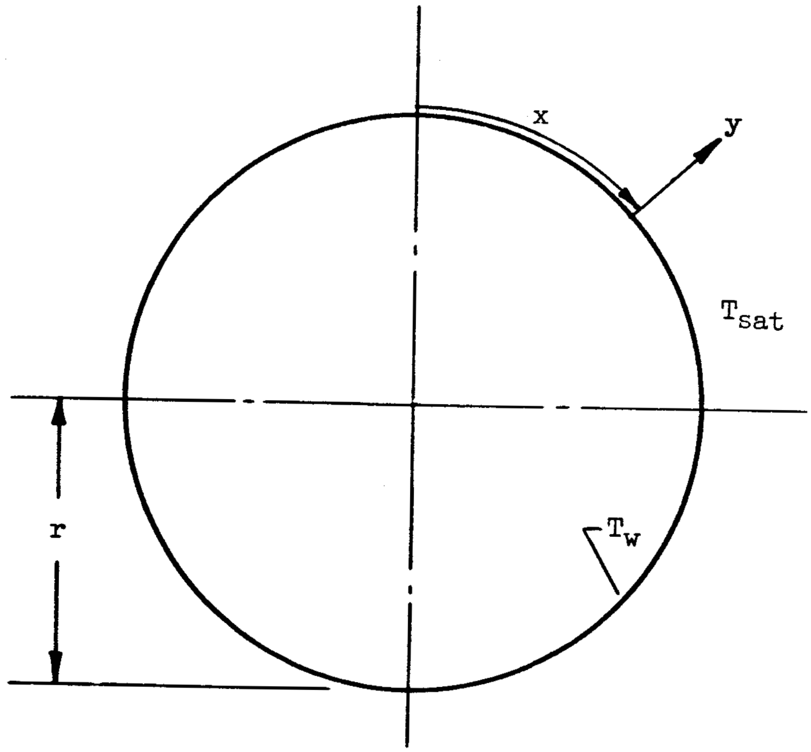

Fig. 1. - Geometrical configuration and coordinates.

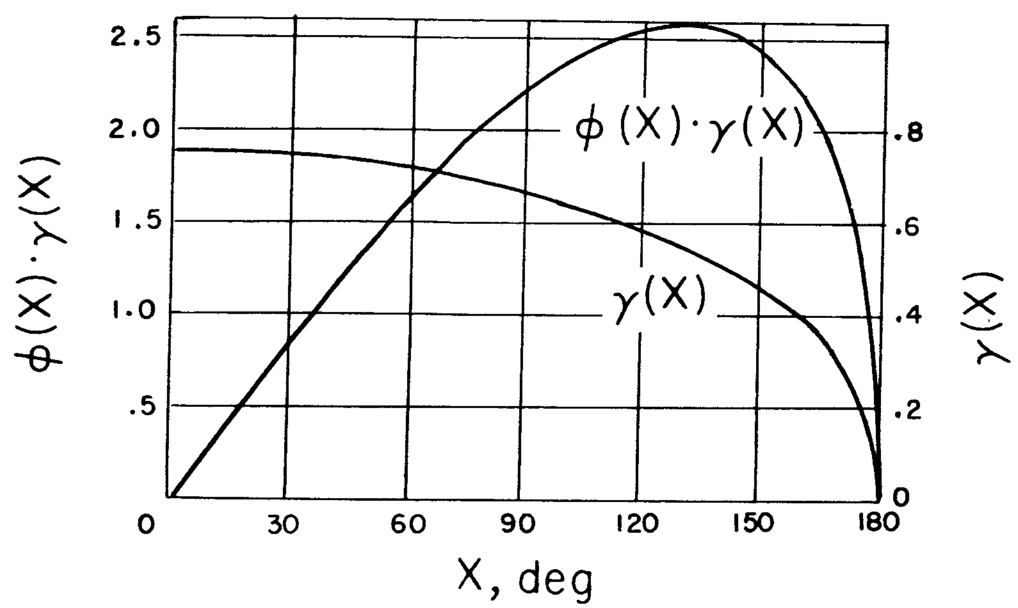

Fig. 2. - The functions $\dot{\gamma}$ and $\varphi\gamma$ (reproduced from fig. 17, ref. 2).

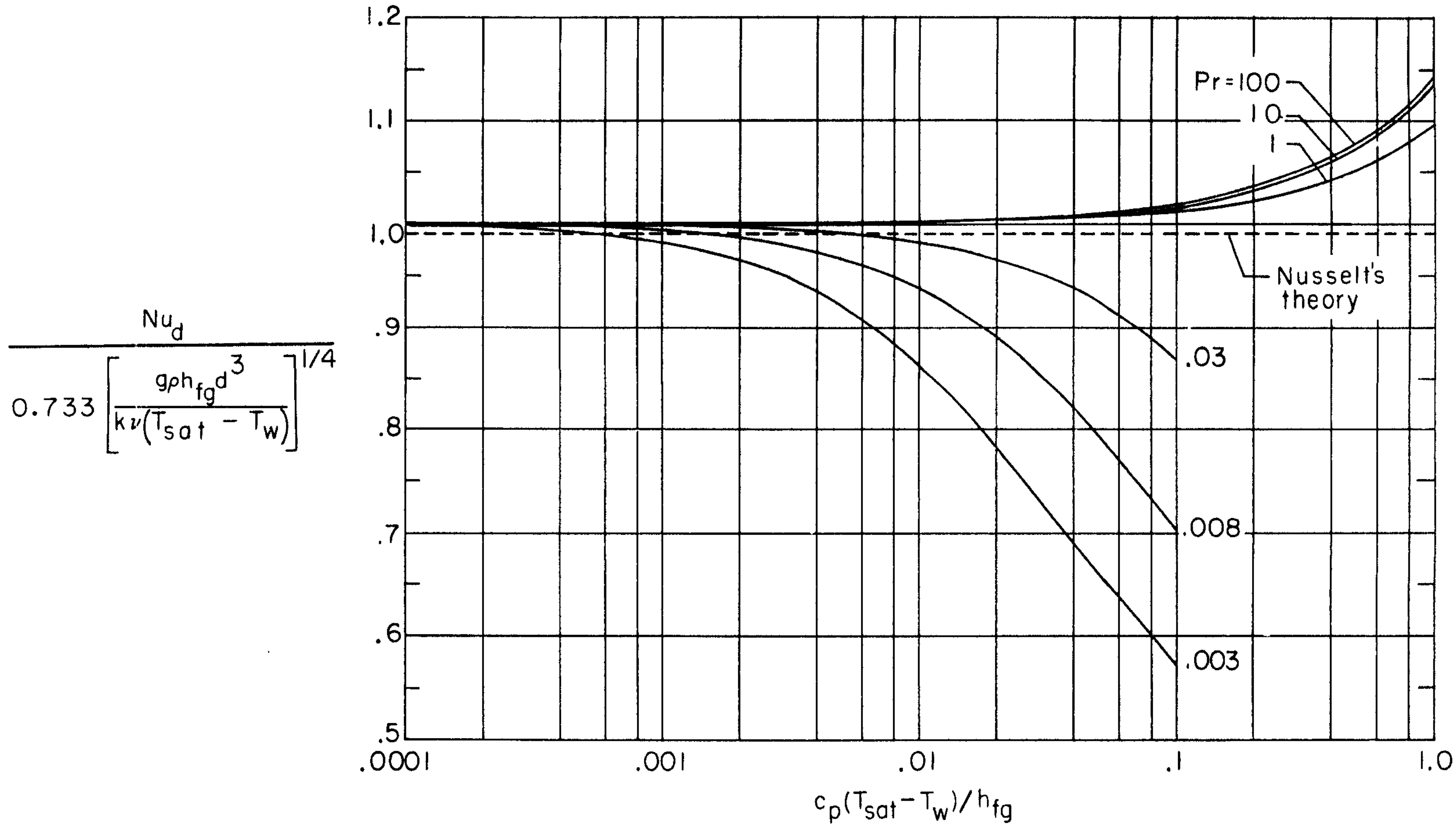

Fig. 3. - Heat transfer results.